CHAPTER 12

Special Structures and Structural Details

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CHAPTER 12

Special Structures and Structural Details

Introduction

1 This chapter completes discussion of the design of parts of the ship's structure and includes detailed or unusual aspects which have not been covered in earlier chapters. The remaining chapters of this publication discuss attributes of the structure which need to be considered in particular circumstances and may apply to any part of the ship. The contents of this chapter therefore may seem a rather random selection, but they are all important topics in their own right although, as will become clear, there is much more detail available to the designer on some aspects than on others.

2 The discussion commences with the design of specific parts of the structure of a ship which are not necessarily part of the hull but are integral with it, such as masts. It then continues with the treatment of structural details, such as holes and connections, but in general terms indicating how the details interact with the overall design; actual design of typical details is covered in other publications, for example for RN warships NES155 Part 1 for general details and NES150 for seatings. Finally, the design of some appendages is included.

Flight Decks

3 The design loads on flight decks are described in Chapter 4 where it is suggested that a logical process is to design the structure for the worst load that an aircraft can apply, which will usually be in an accident or crash on deck. In this case the oleo ultimate load with a suitable margin to take account of dynamic effects if appropriate. If the area concerned is for parking aircraft only, then it will be sufficient to use the aircraft all-up-weight (AUW) distributed to all its wheels, again with a margin for tie down forces, ship motion and wind effects. In either case the important design criteria are that there should be no chance of the deck being ruptured in an accident so that fuel can spill into a space below and cause a fire hazard, and that the distortion of the deck should never be so great as to hinder the passage of aircraft or weapon trolleys, or be unusable for further flying operations. A further consideration is that the deck might, in future, be required to operate heavier aircraft than originally specified and a margin for this eventuality may be appropriate.

4 The design method proposed here assumes that the loads have been calculated in accordance with Chapter 4 and is based on the work of Jackson and Frieze (1980). There is a dearth of experimental data on the loads involved when an aircraft lands on a moving metal structure, most relevant research having concentrated on undercarriage design for landing on rigid runways. Consequently, there are some restrictions to the use of the following method, and in particular it applies strictly only to *vertical* landings, or at least only to those with a small horizontal velocity component. Reference should be made to US Navy methods (for example NAVSEA data sheet DDS 130-1 (1957)) for loads and structural design to withstand conventional fixed wing aircraft landing. Another uncertainty is the distribution of load over twin undercarriage wheels, especially when one wheel is located on a stiffener and the other is near the centre of a plate panel. An estimate of the distribution is given below but it is not supported by strong experimental evidence.

5 The design method separates the calculation of plate thickness from that of stiffener scantlings since the former uses the elasto-plastic method of Jackson and Frieze (1980) while the latter is a wholly elastic analysis. For the very rare accident condition assumed it may be significantly conservative to design the stiffeners elastically, but there are currently no theoretical or experimental data available to justify any other approach.

6 If the flight deck is part of the hull girder then it is necessary to design the structure to withstand the wave and environmental loads first. Deck landings can only be carried out in reasonably benign weather conditions so landing loads may be treated independently of wave loads and a structure which has been designed for wave loads need only be checked to ensure that it is strong enough for landing loads. Aircraft parking loads, however, if they are sufficiently close together (see Chapter 4, paragraph 32), should be applied concurrently with wave loads and this combined condition must be allowed for as described in Chapter 7. Remember also that the design method described here allows for plastic deformation of the plate, which could affect the assumptions on severity of initial deformations used in design of primary structure, and it is probably wise to use the large initial imperfection *plate* curves (data sheets 18.5 and 18.8) although not necessarily the large imperfection *column* curves.

Plating Design

7 Because the design method is elasto-plastic it is necessary to use allowable values of permanent set of the plate. The total permanent set (s) at the centre of a plate panel is the sum of the initial deformation (s_o) due to the construction process, and of the deformation (s_p) due to the applied load. The value of s_p to be used will be that permissible under the maximum applied load which is most unlikely to occur more than once on any single plate panel. Normal operational loads are assumed to cause an elastic response and so there is no requirement for any cumulative deflection to be allowed for, a single maximum load application being sufficient.

8 The allowable value of total permanent set is dictated by the need to move aircraft around the deck, as well as the movement of weapon trolleys, without them having to be manoeuvred 'uphill' to too great an extent, especially in icy conditions. Weapons trolleys also may have a very small ground clearance. Consequently, a value of total permanent set of b/50 is recommended, where b is the smallest plate dimension between stiffeners. However, if there is evidence available to the designer which might suggest that this is too large, or even too small, then the best evidence and advice should be used.

9 Statistical analysis of initial plating distortions has been carried out by Somerville et al (1977) for warships and by Antoniou (1980, 1984) for merchant ships. From the data presented it can be deduced that typical mean plating distortion is of the order of b/120 and is recommended for use here in the absence of more specific data. This value of s_o associated with a value of b/50 for s permits a permanent deformation (s_p) due to the loading load alone of b/85.

10 For plating subject only to parking loads it may be desirable to limit the total permanent set to a value of b/100 to make wheel movement easier. This is little more than the expected initial deformation on construction and so under these conditions it would be necessary to design the plating elastically. However, a practical limit to permanent set due to an extreme parking load is probably about b/150, giving a total permanent set of approximately b/65.

11 The elasto-plastic behaviour of a plate under 'patch' loading (loading over an area less than the plate size but more than a point) is characterised by three non-dimensional parameters:

(12.1) $$C_p = PE/(b^2\sigma_y^2)$$

(12.2) $$C_{sp} = (s_p/b)\ \sqrt{(E/\sigma_y)}$$

and

(12.3) $$ß = (b/t)\ \sqrt{(\sigma_y/E)}$$

where P is the applied load (not pressure) modified if necessary as described in paragraphs 12 and 13 below, and other parameters are as previously defined above and in Chapter 7, ß being the plate slenderness ratio. Tyre 'footprint' dimensions are b_t for width and a_t for length with b_t parallel to b. Plating design curves are given in data sheets 18.75 to 18.77 for three different values of C_{sp} where, for a particular value of C_p and b/t_t, the required value of ß and so plating thickness (t) can be read off. For values of C_{sp} and b/t_t between those given, interpolation is adequate. The thickness derived must be rounded up to account for rolling tolerances as defined in Chapter 5, and a corrosion allowance made if considered necessary.

12 The plating design curves assume the load is applied through a tyre which has a rectangular footprint with an aspect ratio of 2, that is $a_t/b_t = 2$. If no details of the footprint are available this assumption is adequate with b_t equal to the nominal tyre width. If the tyre is known to have a different aspect ratio then the load P should be increased by a factor $(2b_t + 1.1b)/(a_t + 1.1b)$, which has been derived from information obtained from Lloyd's Register for the development of their Rules.

13 When the undercarriage is twin wheeled it may be necessary to derive an equivalent load. The load may be equally divided between the two wheels if $(b_t + w)/b > 0.6$ where w is the spacing between the wheel centre-planes, that is the design value of P is halved, otherwise the load on one wheel is increased to allow for uneven distribution of the load due to landing attitude and also for the wheel spacing relative to the plate width. The design load P should then be divided by an 'equivalent load factor' taken from data sheet 18.78 for the particular values of w/b and b/b_t, interpolating as necessary.

14 Prior to undertaking the plate thickness checks detailed above, the orientation of the aircraft on landing should be considered. If the flight deck overall dimensions permit fore and aft *and* athwartships landings then the checks must be performed for both directions. In each case the dimensions b and b_t are to be taken as parallel to the axle, that is at right angles to the direction of landing, and care should be taken if using the expression in paragraph 12 to use the values of b, t_t and a_t relevant to the direction in question.

15 For a predominantly longitudinally stiffened deck it is unlikely that the parameters will exceed the limits of the design curves. However, for landing at right angles to the principal stiffener direction the value of b/b_t may fall outside the scope of the curves. In this event limited cross plotting and extrapolation is permissible, but if the value of b/b_t is more than about 8 then it is probably best to change the stiffening arrangement.

16 The above procedure applies specifically for plating in landing areas. For parking areas, if a permissible permanent set is specified, then the same procedure may be used. Otherwise an elastic design method is appropriate (see Chapter 10) and the tyre should be taken as both a point load (P) and a pressure load (P/ab) with design being for the worst case.

Stiffener Design

17 The stiffeners of the flight deck should be analysed elastically as a grillage under a point load as described in Chapter 10. However, because there will be a number of potential load points and it will not be immediately obvious which is the worst, it will usually be best to use finite element analysis, modelling the grillage such that the load point can be readily changed. The worst conditions are likely to be with the load in the middle of a beam span between the supporting stiffeners at right angles so it will be necessary to model the critical beams with nodes at mid-span and with at least three nodes between supports. To decide on which beams the load should be applied it will be necessary to consider the likely operational conditions, that is the boundary within which an aircraft can touch down a single wheel, or where it is intended to be parked. Unusual, but possible, operating conditions should not be forgotten. Note that for landing a single load point is appropriate, while for parking all the wheel loads should be applied together (and possibly for a number of aircraft at once, but see Chapter 4, paragraph 32).

18 The grillage analysis will not allow directly for the plating, and an effective breadth is required. Because of the nature of the load, the effective breadth that would be suggested by Chapter 10 can be a little pessimistic and the following rule is suggested here:

(12.4a) for $b/t < 100$ $\qquad b_e = b\,(1 - b/200t)$

(12.4b) for $b/t \geqslant 100$ $\qquad b_e = 50t$

However, the section modulus of the stiffener and associated plating is not very sensitive to the value of b_e and so the lesser of b/2 or 40t may still be used.

19 The load on a stiffener due to twin wheeled undercarriage needs modifying in a similar way to that for a plate. Data sheet 18.79 gives an equivalent load factor for the stiffener for values of w/b, but is only appropriate when the wheel is in the centre third of a beam span with one tyre on the beam. Otherwise the full load P should be used even if it is in reality slightly offset, and this condition should always be assumed in any case of doubt.

20 Using the landing loads from Chapter 4, modified if necessary as described above, the stress in any stiffener in a landing area should not exceed the yield stress. All margins are incorporated into the design load and so no further stress factor is required. However, a stress margin of 1.2 against yield is recommended for parking and aircraft working areas outside the landing area to guard against the remote possibility of local fatigue damage due to repeated aircraft movements.

21 Checks should also be carried out against the possibility of stiffener tripping as described in Chapter 7, paragraph 35, when the margin between applied bending stress and critical tripping stress should be at least 4.

General Design Requirements

22 Remember to ensure that the load bearing capacity of all supporting structure, such as the bulkheads and pillars, is adequate to take the load transferred from the flight deck.

23 Care should be taken that the principles discussed in Chapter 14 for design to minimise vibration are fully accounted for, because flying operations generally take place at a high ship speed when exciting forces are high, and any local vibration in a flight deck can be very irritating, if not dangerous.

24 Special attention should be paid to local structural details such as stiffener connections and connections between the deck and supporting bulkheads. Landing loads are impulsive and any stress concentrations can cause fatigue damage early in the life of a structure if care is not taken.

Mast Design

25 It is notable that few of the published texts on ship design say anything about the design of masts, and there are none that appear to mention mast structure. Lloyd's Rules (Section 3, Chapter 9) have a short section on masts but it only includes qualitative remarks, mainly about support structure. However, instructions do exist in other navies' design documentation, notably from Canada and the USA, and it is these from which the following guidance is drawn.

26 There are three types of mast which are of general use in warships, the pole mast, the lattice mast and the plated mast (*Figures 12.1a-c*). Of these the latter is of common use only in European navies, while North American ships usually have lattice or even pole masts. Consequently, the available instructions do not include much on plated masts, although the general requirements will be similar for all types. A pole mast has the advantage of light weight and simplicity but has a limited carrying capability and a tendency to low stiffness, and hence

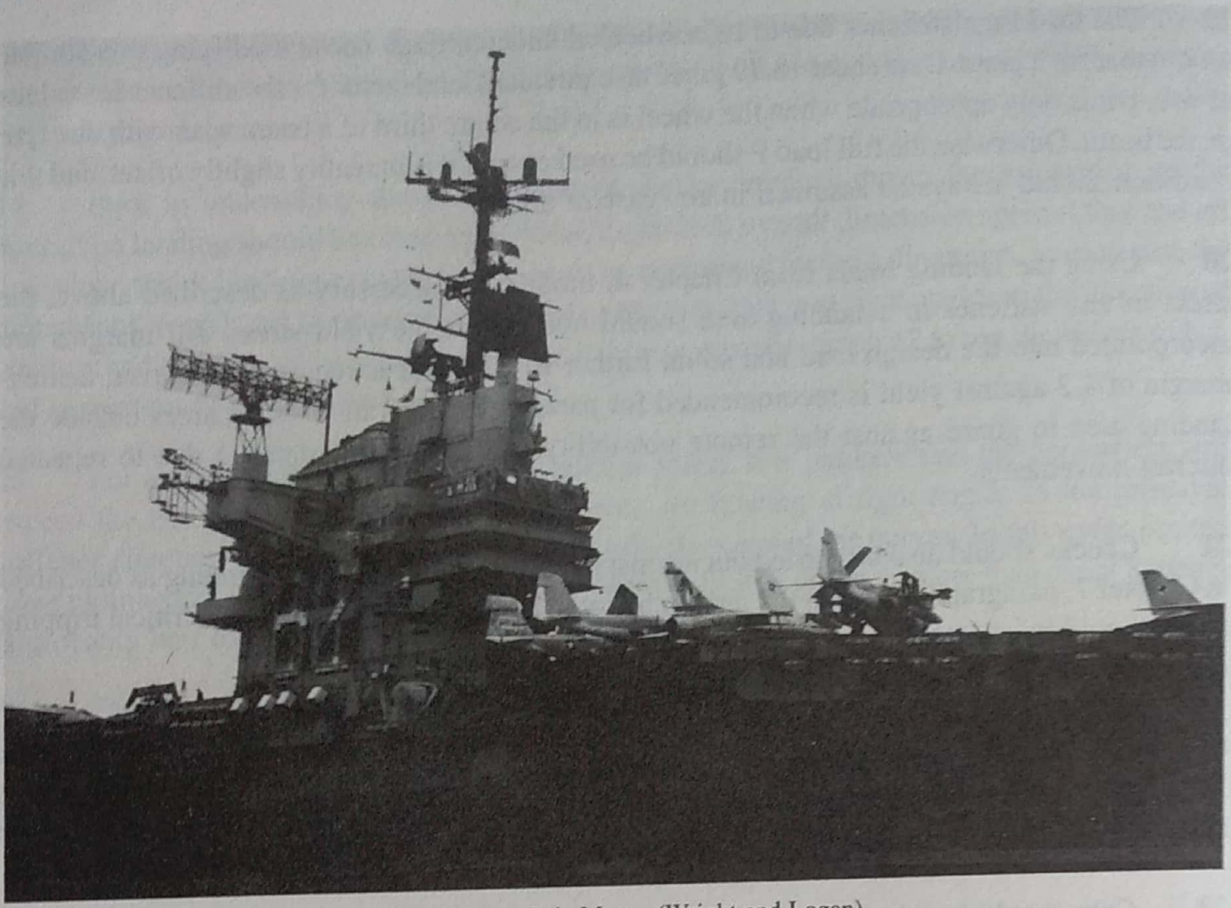


Figure 12.1a Pole Mast (Wright and Logan)

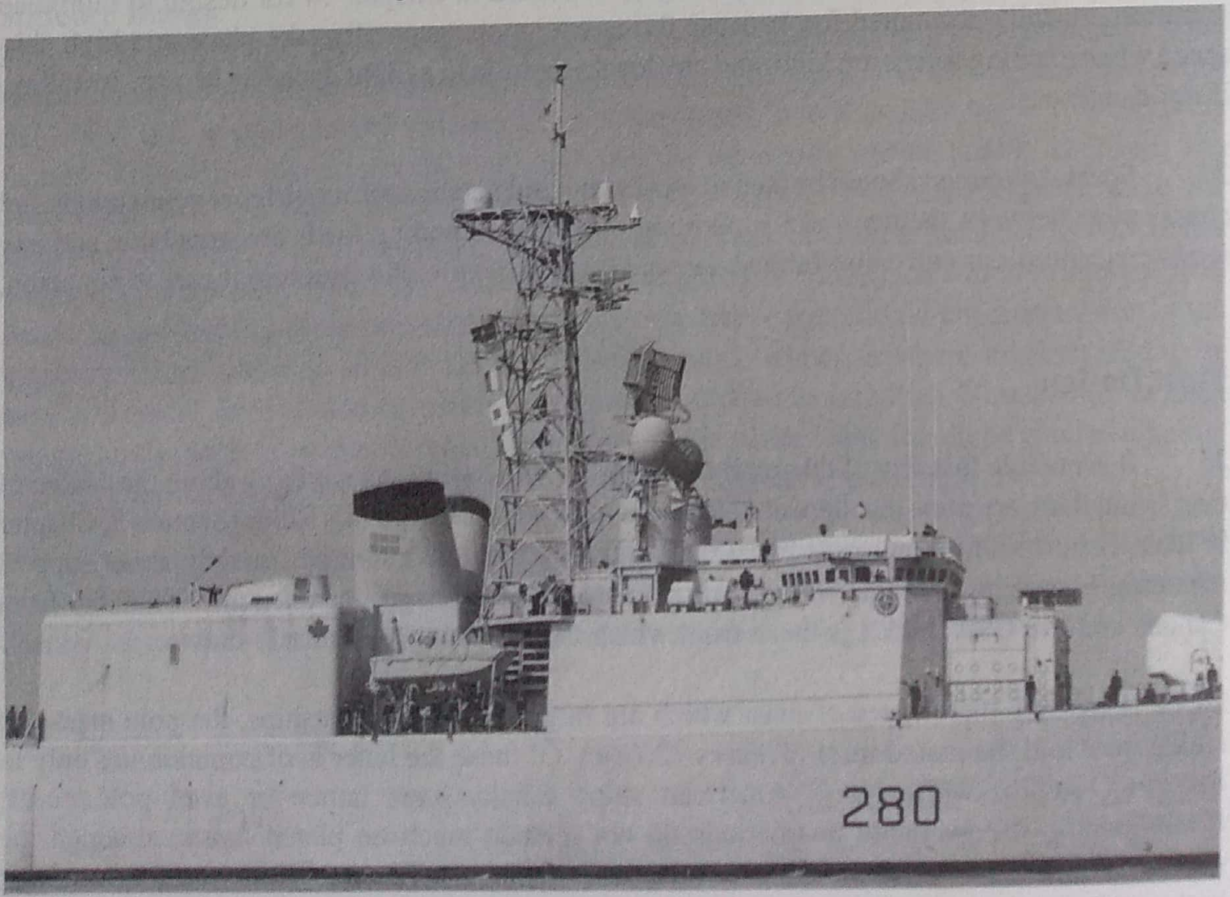


Figure 12.1b Lattice Mast (Wright and Logan)

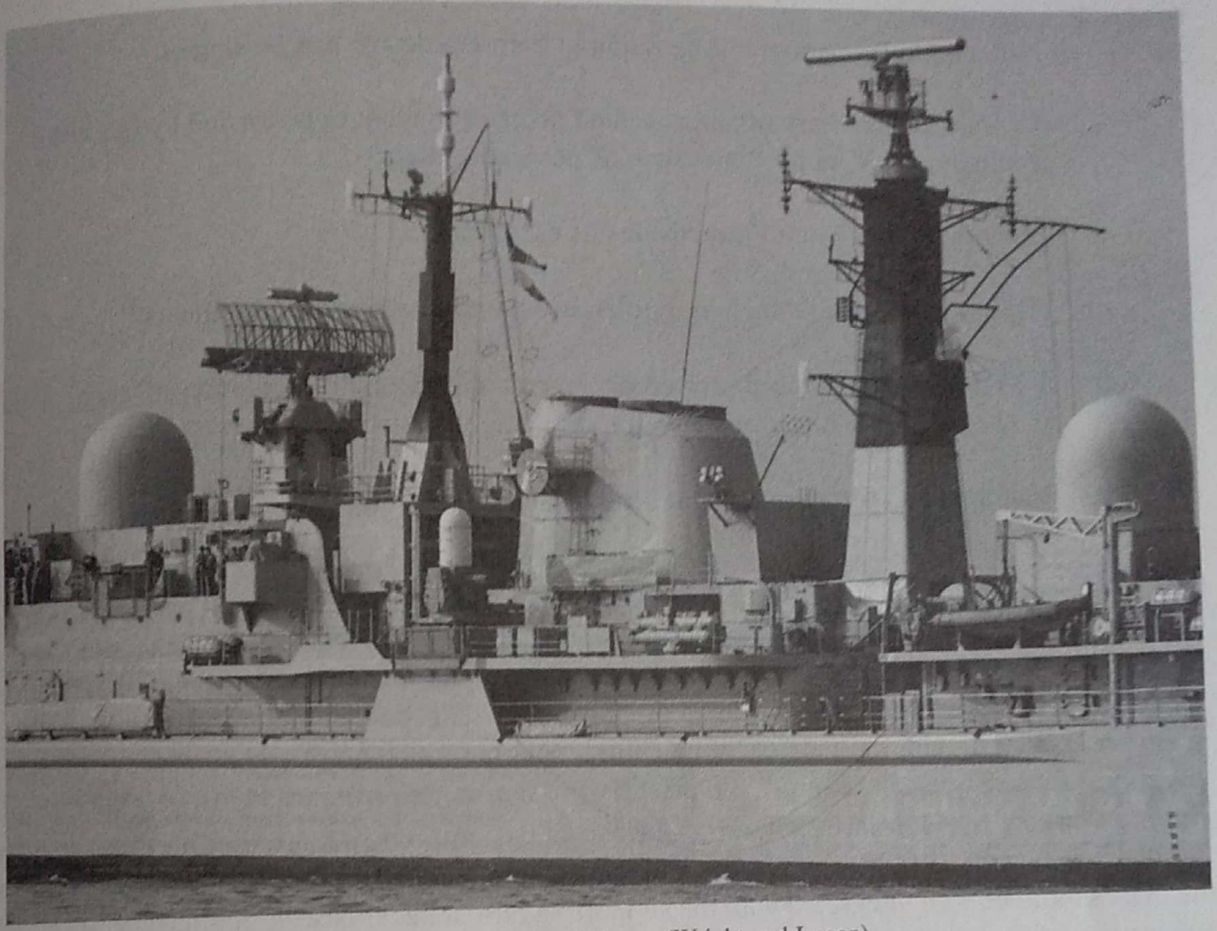


Figure 12.1c Plated Mast (Wright and Logan)

large vibration amplitudes under normal wave and machinery excitation (although pole masts may be stayed to reduce their deflections). A lattice mast overcomes this problem by spreading the base area and increasing the bending stiffness, but it then becomes difficult to clean, preserve and maintain with many inaccessible corners liable to hold water and to corrode. Additionally, it is less transparent to radar due to the many angled corners (see Chapter 8) and presents a significant shadow for the ship's own radar. Consequently, the Royal Navy now has a tradition of using plated masts, which overcome some of the disadvantages of lattice masts but they are even less transparent to the ship's own radar and create a larger shadow area. However, for a mast only required to support a small load and where stiffness is not so important, the pole mast should not be dismissed.

27 Another possible configuration is that of a combined mast and funnel, or a 'mack' (mast + stack). This will be similar to a plated mast except that part of the inner space is used for machinery uptakes and may, additionally, be cantilevered out from a deckhouse. The method of design will be the same as for a plated mast but even more attention will need to be given to access for preservation.

Design Requirements

28 The following information will be required before a design can be started:

a. A complete inventory of antennae and other equipment to be carried by the structure (excluding very minor items such as navigation lights).

b. The weight and overall dimensions of each item.

c. The mounting details including tolerances) and clearance requirements.

d. A sketch showing the preferred arrangement and relative location of each of the items, together with details of minimum heights above deck or waterline.

e. Requirements for separation of sensors and arcs of fire.

29 A mast must be designed to withstand the loads defined in Chapters 3 and 4 caused by wave action and by air blast and underwater shock, while at the same time deflecting by an amount which is consistent with the sensor design requirements, in particular the relative deflection between the sensor and weapon or ship datum. This deflection may also be caused by vibration excited by hull motions or by machinery and the frequencies and amplitudes of these vibrations should be such as not to interfere with sensor performance. Further guidance on vibration effects is given in Chapter 14, but particular requirements are that the mast natural frequencies must be as far removed from hull natural frequencies as possible and that the base of the mast should be well integrated with the supporting hull or deckhouse structure. The lower the supporting stiffness, the lower will be the mast resonant frequencies and the higher the amplitudes. A mast should always penetrate or be attached to at least two decks at which it is held laterally, and at the same time at least one face must be directly over a main transverse bulkhead. The other faces should also be sufficiently well supported vertically to reduce 'rocking' motion.

30 The mast should also create minimum interference to the operational performance of radars and other sensors on itself and on other masts. Safe access should be provided, either inside or outside the mast, to all the sensors as well as to cabling, wave guides and to the structure for maintenance, while at the same time ballistic protection should be provided for cables and wave guides.

Design Margins

31 With the ever increasing demands being made on top-side arrangements, and a larger number of sensors competing for prime positions, the designer would be unwise to assume that the list of items obtained above (paragraph 28a) will not be added to over the life of the vessel or class. Margins introduced must take account, therefore, of the operational requirements for the ship, likely missions and possible changes in the future, the expected life of the ship and any relevant weapon development programmes. Consequently, it is difficult to make any firm recommendations for design margins and each case must be considered on its merits, however, the following points should be borne in mind:

a. The combat system suite is more likely to be modernised than any other part of a ship and so a growth margin will be needed on the mast which is greater than that for the ship as a whole.

b. The application of growth margins will require not only weight and centre of gravity, but also space, height and windage area. It may also be desirable to allow for the addition of more appendages (yards) at a later date. The US Navy requirements suggest a factor of safety against yield stress 25% greater than would normally be necessary, to allow for growth.

Material Selection

32 Masts are subject to a continuous cycle of loads due to ship motion and to wind and so there is a high risk of fatigue damage. Additionally, the structure is high in the ship and so low weight is important, while a high stiffness is necessary to reduce vibration problems. Difficulty of access except in harbour also suggests that low maintenance is an important consideration. Generally, these requirements are incompatible, but the fatigue requirement will probably rule out the use of aluminium, while the need for high stiffness militates against conventional glass reinforced plastic. While steel is likely to be the first choice, it is heavy and requires good preservation if maintenance is not to be a burden, and so some interest is now being shown in high performance composite materials utilising high stiffness fibres. Composites should certainly be considered at least for the upper parts of the mast and for yards, and further information on its advantages and disadvantages is given in Chapter 5.

Design Approaches

33 The following discussion assumes that a plated mast is to be designed; for a pole or lattice mast reference should be made to the US Navy or Canadian Navy standards respectively. A plated mast can be taken as a cantilever structure which is fixed at its base with some rotational constraint. Clearly, the higher the restraint the stiffer will be the mast and the higher its resonant frequencies, and so again, clearly a mast should never the positioned in the middle of a plate panel where the restraint is dependent on plate bending stiffness. It is recommended above (paragraph 29) that a mast structure is supported on a bulkhead vertically and restrained laterally at least two decks. Consequently, the design problem reduces to that shown diagrammatically in *Figure 12.2*

34 With the loads known from Chapters 3 and 3, the best way of commencing the design is to estimate the maximum bending moments both athwartships and fore and aft using simple beam theory. From these the minimum section moduli of the mast at the two supporting decks can be deduced, aiming for a factor or at least 2 against yield stress (plus any growth requirement) so as to lead to adequate stiffness when the structure is analysed. From the required aspect ratio (square is usually a good start if no further guidance is available) a structure can then be synthesised using a process similar to that proposed for the hull section in Chapter 7 (paragraphs 5-6) but simplified here because the section is rectangular and a bending moment is known for both planes of

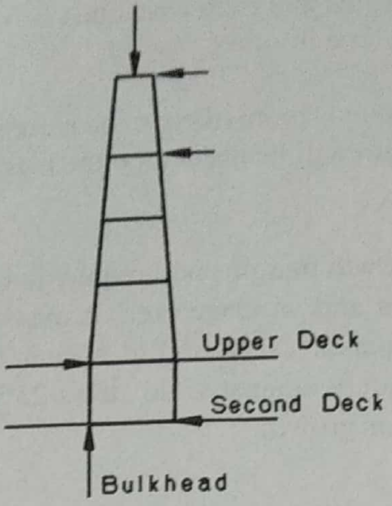


Figure 12.2 Diagrammatic Arrangement of Typical Forces on a Mast

bending. The section of the mast may be tapered upwards matching, as far as possible, the reducing section modulus to the reducing bending moment, and at critical sections calculating the buckling strength of the structure (it is likely to be vertically stiffened, with horizontal diaphragms analogous to transverse bulkheads in the hull) ensuring an adequate margin of strength. As it is unlikely to be worth carrying out an ultimate strength analysis of the mast, the individual panels and vertical stiffeners should have a margin of at least 1.5 against critical buckling stress under the worst *combination* of loads to be expected (usually inertial loads due to ship motion, gravity and wind loads). Fatigue life should also be checked, although advice may need to be sought from specialists as there is little information available on the load history of masts; for design in steel a maximum stress *range* (tensile to compressive) of 200 MPa is probably a good aim to begin with.

35 Finally, a dynamic analysis should be carried out to ensure that the vibratory and transient shock environment is acceptable to the items carried on the mast. If the structure is still taken as a tapering beam, a Myklestad-Prohl method may be used (see Myklestad (1944), Prohl (1945) and Todd (1961), as well as Chalmers (1988(a)) for an application in a different context) although an assumption will need to be made on the lateral and vertical stiffness of the restraining decks and bulkheads, and it will be desirable to err on the conservative (less stiff) side. Otherwise it will be necessary to carry out a finite element analysis, which should then include a portion of the supporting structure out to the nearest transverse and longitudinal bulkheads (or ship's side) and down through all the decks onto which the mast is stepped. This analysis will give the most reliable predictions providing that the model is well thought out and adequately representative, ensuring that each part of the structure which provides a critical load path is modelled in detail.

Seatings

36 The design of seatings is covered in detail in NES150 and there is, therefore, no need to present more than the main principles here. The main purpose of a seating is to support an item of machinery and equipment under all normal operational loading conditions, to maintain alignment if necessary and to provide protection from hull borne vibration and underwater shock accelerations. At the same time the seating may provide a barrier to vibration or noise transmission from the equipment being supported. Consequently, the initial design problem will stem from dynamic considerations rather than static loads and the following features should assist in achieving an adequate solution:

a. The seating should be well integrated into the hull, deck or bulkhead structure to provide maximum strength and stiffness without too heavy scantlings or significant discontinuities and stress concentrations.

b. The seating should be as low as possible and certainly lower than it is wide.

c. Load paths between the mounting points and the ship's main structure should be as short and direct as possible, and loading parts of the seating in bending should be avoided.

d. The seating should be braced to give good stiffness in shear and torsion.

e. Where appropriate, and especially if a set of machines is required to be held in close alignment with each other, it is usually more efficient to mount a number of items on a single seating.

f. Notwithstanding the remarks on stiffness above, it may sometimes be desirable to use a seating as a 'mechanical fuse' to protect equipment from shock loading, that is, the seating absorbs energy through plastic deformation rather than transmitting it elastically. Clearly this is not suitable if alignment is important.

g. Wherever possible the seating design should avoid inaccessible regions where corrosion can progress unseen.

37 Note that seatings should be designed to be significantly stiffer than the mountings that are fitted between them and the equipment (usually at least 10 times as stiff) so that there is no interference with the noise or vibration isolation efficiency of the mount. To achieve this requirement it is unlikely that cantilevered seatings will be acceptable except for very small lightweight items.

38 Because of the uncertainty of loading, especially under shock, except when a seating is designed deliberately to yield, it is recommended that no part of the seat can experience direct or bending stresses greater than one third of yield stress, or shear stresses greater than one half of shear yield under the worst expected load. Brittle materials should always be avoided as these respond unpredictably under shock loading.

Break of Forecastle

39 The 'break of forecastle', which is usually a break down from 1 to 2 deck and may actually be well abaft amidships, presents a major discontinuity in the hull girder similar to the ends of a large superstructure block. As such it is best avoided if at all possible. If there is no choice but to have one then very great care must be taken in the integration of the structure in way of the break and the break down should be as far from the region of high wave bending moment as possible.

40 Various structural arrangements have been tried in the past, most recently in the RN on the DARING and BLACKWOOD classes; none are now extant and no arrangement has been found which is entirely satisfactory. The problem reduces to how to carry the in-plane load from the upper of the two decks to the lower one. Ideally, a sloping ramp would be incorporated but such an arrangement would be incompatible with layout and inefficient in space. The next best alternative is to provide material at the sides of the upper deck to transfer the load into the ship's side which then has to be well stiffened to cope, and all unnecessary stress concentrations removed from the region. A recommended style of design is shown at *Figure 12.3* based on work at the Naval Construction Research Establishment (now the Defence Research Agency, Dunfermline) in the late 1950s.

41 In the design in *Figure 12.3* the load in the upper deck is diffused into the rest of the hull through the side plating. The profile of the side plating in way of the break is especially important and it is necessary to stiffen the free edge against instability in compression. In addition, it is

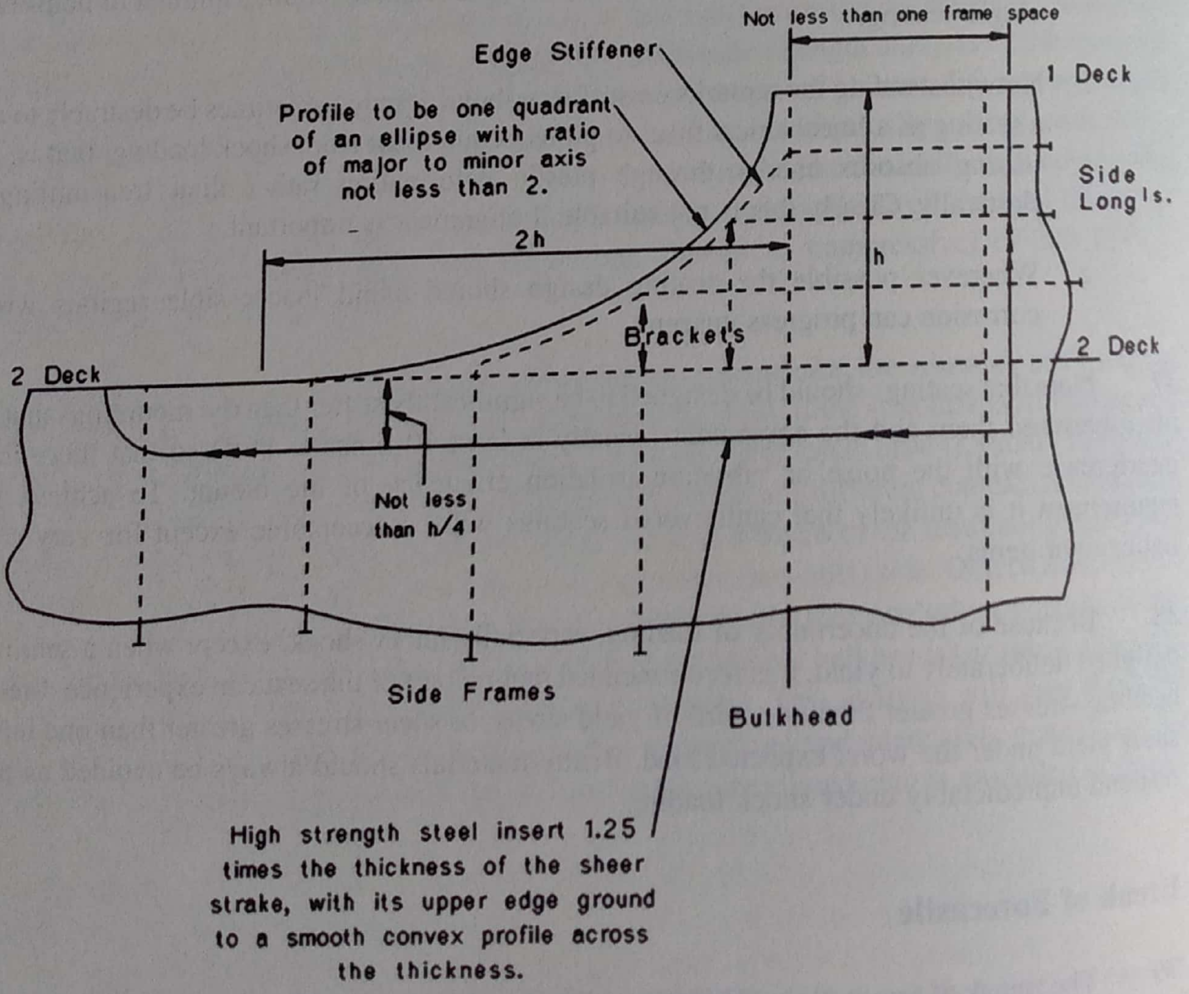

Not less than one frame space
Edge Stiffener
1 Deck
Profile to be one quadrant of an ellipse with ratio of major to minor axis not less than 2.
Side Long'ls.
2h
h
Brackets
2 Deck
2 Deck
Not less than h/4
Side Frames
Bulkhead
High strength steel insert 1.25 times the thickness of the sheer strake, with its upper edge ground to a smooth convex profile across the thickness.

Figure 12.3 Suggested Arrangement for Break of Forecastle

advisable to fit a bulkhead at the end of the upper deck which extends at least one deck below 2 deck so as to transmit the vertical compatibility forces. The requirements of paragraphs 60 to 62 for connection details must be strictly adhered to.

42 It is evident that the curvature of the side plate edge where it rises from the lower deck should be made as gradual as possible. The upper part of the slope where it approaches the forecastle deck level carries lower stresses and higher curvature can be accepted here. A suitable profile is formed by a quarter of an ellipse with semi-axes chosen to give a radius of curvature at the bottom of the slope of at least twice the deck height reducing to be the same as the deck height at the top.

43 However gradual the curvature, some concentration of stresses is inevitable and only if there is no alternative should there be welds ending in the side plate edge. The plate forming the side should be as long as possible, approximately of the size shown in *Figure 12.3*: If plates of these dimensions are not available then a portion containing at least the lower two thirds of the slope plus a distance equal to half the deck height abaft the bottom of the slope should be cut from one plate.

44 The material used for the side insert plate should be of high strength and toughness and Q1(N) steel to NES736 Part 1 or an equivalent is recommended. Although only a very limited reduction in strain can be achieved by thickening the side plating in way of the break, some increase in thickness has the advantage of increasing the margin against instability in compression as well as providing some insurance against bad workmanship and corrosion. However, too great an increase will lead to discontinuities at the boundary, so it is recommended that the thickness should be about 25% greater than the thickness of the sheer strake in that region, but if the precise thickness cannot be obtained in the recommended material it is more important that the right material is used.

45 The top edge of the side plate should be cleaned of all tool and burning marks and finished to the elliptical profile with a rounded edge as shown in Figure 12.3. No drainage holes should be cut in the side in way of the break, and attachment of fittings to it should be avoided if at all possible.

46 It will be necessary to stiffen the side plate to prevent buckling, and a good arrangement is to use Tee or angle sections supported at intervals by stiff vertical athwartships brackets attached to the deck over each deck beam. The stiffener should run in a series of straight cords between the bracket tops close to the edge of the plate but at a minimum distance of 40 mm from it. The longitudinals in the forecastle side should continue aft and terminate at suitable brackets as shown in Figure 12.3. Abaft the after end of the stiffeners there will be a small area of unsupported plate edge, and the height of this should nowhere exceed 20 times the plate thickness.

47 To reduce twisting and possible tripping of the edge stiffener it is desirable that the stiffener section should be as small as possible, consistent with its purpose. The flange should be tapered to zero immediately aft of the last supporting bracket and the last 300 mm or so of the web should be curve downwards to meet the deck at a beam or onto a small insert plate of 1.5 times the deck thickness. The free edge of this part of the stiffener should also be ground smooth with special care being taken over the welding in the area. This curvature of the stiffener allows transmission of forces gradually from it into the side plating.

Efficiency of Short Decks

48 In estimating the effectiveness of structure in taking primary loads as discussed in Chapter 7, decks below a certain length were disregarded. However, such short decks will still take some longitudinal load and must be designed accordingly. The percentage of primary longitudinal bending stress in the deck as a function of distance from a free edge and the deck breadth can be estimated from data sheet 18.80; this stress is the *average* across the deck section, so as the strain (and therefore the stress) will be the same as the primary strain at the ship's side the stress at the centre will be a little less than that found from data sheet 18.80. Nevertheless, for design purposes the average values should be used except for, say, the outer two longitudinal stiffener spacings where the ship side stress is applicable. Note that any secondary stresses will need to be added to the primary stresses; information on the estimation of secondary stresses is given in Chapter 10.

49 For short lengths of inner bottom the value of breadth B for input to data sheet 18.80 should be taken as the spacing of the longitudinal plate floors supporting the bottom, and in consequence the efficiency of inner bottom structure in this context is always likely to be near 100%. Note that this does *not* imply that effective width of plate is anywhere near 100%.

Holes in Structure

50 Holes in any highly stressed structure should be kept to a minimum, but a number are inevitable. Where they exist they will cause the stress to be increased locally, both due to the reduced amount of material carrying the load and due to the need for the load path direction to change so creating biaxial or triaxial stress effects. The loss of material, if it is significant, has to be compensated for by additions elsewhere, for example in a coaming, but the other effect can only be minimised by good design and avoidance of sharp corners.

51 Holes may also affect the buckling strength of the structure but there is very little information published in this area. As a general rule, if the hole is designed so as to lead to acceptably low stress concentrations for limiting fatigue damage, then its effect on buckling can be ignored, but the designer should bear the problem in mind for unusual hole configurations.

52 To assist in estimating the level of stress concentrations, there are a number of theoretical solutions for idealised holes in simple stress fields based mainly on plane stress analysis or on conformal transformations and mapping. Examples are given, in particular, in Roark 'Formulas for Stress and Strain' and in Volume 7 'Stress Concentrations' of the ESDU Structures Data Sheets. All such solutions are for isotropic materials; for anisotropic materials the stress concentrations may be higher and a solution can only be estimated by using finite element analysis, as indeed would be necessary for unusually shaped or very large holes.

53 The factors which govern the importance of holes in a stress field are:

a. The position of the hole in relation to the applied stress.

b. The shape of the hole.

c. The size of the hole.

d. The reinforcement provided.

For a. there is little to be said except that a hole should be placed in as low a stress field as possible, although frequently design considerations make this difficult. Otherwise circular holes are preferable as they conform more readily to the ideal stress flow and are easier to analyse reliably. Rectangular holes are, however, the most common and in recent years the size of these has been increasing, especially in container ships where they can no longer be treated as holes. The problem there has become one of designing the remaining surrounding structure to provide adequate primary stiffness and strength, particularly in torsion. Research in the 1960s led to a number of publications in this area, for example Richardson and Ossowski (1965), Williams (1967) and Bell and Richardson (1967, 1968). Conclusions from this research were that in a *uniaxial* stress field parabolic or elliptical corners reduce the stress concentration by about 10% compared with circular corners, and that when there are adjacent holes of similar size aligned in the direction of the maximum stress the stress concentration reduces when the distance between the holes in less than 1.5 times the width of the holes.

54 The optimum reinforcement for small rectangular holes is discussed by D'Archangelo (1964). An unreinforced hole will have a stress concentration of between 3 and 5 at a radiused corner, and the best that can be achieved in reducing this by inserting thicker material around the hole is about 30%, when 60% of the hole material is replaced in the insert. Replacing less than 30% of the hole material is relatively ineffective and is not worthwhile. For circular holes the effect of increasing local thickness is shown in data sheet 18.81 for uniaxial tension and pure shear; for these cases the maximum stress is at the edge of the hole up to the limits shown, beyond which the maximum stress occurs at the junction between the parent plate and the insert and remains nearly constant with increasing insert thickness. There is little to gain from increasing the insert diameter more than 2.5 times the hole diameter and in practice, to make the best use of material, the edge of the insert should be tapered with a slope of not more than 1:4; notch tough material for the insert is also desirable. For large holes it is better to fit inserts at the points of high local stress concentration only, rather than all round. A 50% increase in thickness will give a stress concentration reduction of about 20%.

55 Coamings are more efficient than inserts for reduction of stress as the reinforcement is kept near to the hole edge. However, the effectiveness of the coaming falls off rapidly with height and in practice it is best to keep the height less than 7 times the coaming thickness if it is symmetrical about the hole edge, or less for an asymmetric one. Furthermore, a coaming introduces a worse joint category for fatigue than does an insert plate (see Chapter 13), and for this reason an insert plate may be preferred. For circular holes the optimum reinforcement for uniaxial stress is about 80% of volume or 40% of cross section area removed, giving roughly 50%

reduction in stress concentration. For small rectangular holes it is 25% to 40% of cross section area removed, the lower limit being adequate if the corner radii are not less than 1/8 of the hole breadth. For large rectangular holes coamings reduce the mean stress in way of the hole, but only reduce stress concentrations by 10-15%. For small holes the thickness of the coaming is usually dictated by production considerations such as welding distortion.

56 For critical structure or unusual hole arrangements it may be desirable to fit both inserts and coamings. In such cases the stress concentration may be considered reduced by the product of the effects of the two reinforcements. However, this can only be a first estimate and such arrangements will usually be of sufficient importance to merit finite element or experimental analysis, and indeed finite element analysis is desirable for the area around any large hole or group of holes in primary structure.

57 Holes in webs of stiffeners and plate floors should be avoided in areas of high shear stress. In other areas they should be kept close to the neutral axis and made circular or elliptical. Reinforcement in the form of a rider place or web stiffener around the hole will be necessary if the hole diameter is more than half the web depth.

58 Where there are a number of holes to be sited close together they should be arranged longitudinally along the ship and *not* athwartships, so as to remove as little primary structure as possible in any cross section (see Chapter 6 and *Figure 6.1*), and the same idea will apply to any other piece of stressed structure with holes. Where small holes are to be positioned near to large holes then they should, if possible, be in the 'shadow area' as indicated in *Figure 12.4*.

59 Summarising the design considerations for the siting of holes:

a. Holes should be positioned in low stress areas wherever possible, including shadow areas of other openings.

b. The long dimension should be in the direction of maximum stress.

c. Corners of rectangular holes must *always* be radiused, even in secondary structure.

d. Holes must be as small as possible, although it may sometimes be best to combine a concentration of small holes into one larger one.

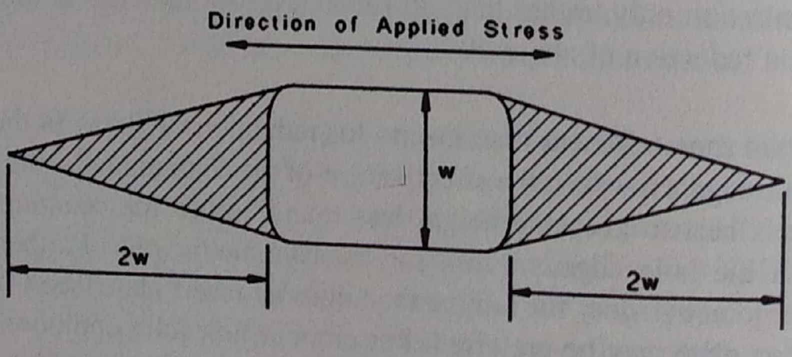


Figure 12.4 Shadow Areas Around Holes

e. In a nearly *unidirectional* stress field adjacent holes should lie in the direction of the maximum stress and should be as close to each other as possible.

f. In a *complex* stress field the distance between hole edges should be greater than the dimension of the larger hole measured in the same direction.

g. Reinforcement should be enough to prevent yield or fatigue failure especially in primary structure. It may be worth fitting higher strength or notch tough reinforcing material.

h. Reference should be made to NES155 Part 1 for further details of recommended hole arrangements.

Structural Connections

60 All structural connections lead to stress concentrations and so should be kept to a minimum and be as simple as possible. At all times the load path through the connection should be borne in mind so as to avoid 'hard points'. Extensive work on stiffener connections was undertaken in the early 1960s following frequent fatigue failures in ships build during the second world war, and is reported by Faulkner (1964) with the practical details being included in NES155 Part 1. Wide ranging reviews of structural details and connections are presented for the Ship Structure Committee in the United States by Glasfeld et al (1977) and Munse et al (1983).

61 A summary of the principal considerations in the design of connections is as follows:

a. Ensure that the connection is necessary; any member connected to a stressed member will itself be stressed and the number of attachments should be minimised.

b. A connection between two components should not weaken either of them to the extent that essential strength or stiffness requirements are compromised.

c. A connection should be stiff enough to prevent any relative movement which would alter the elastic behaviour of the structure.

d. Connections which load the material through the thickness should be avoided as they rely on the through thickness material properties which are often not defined.

e. Connections must be easy to fabricate, inspect and maintain.

f. Alignment of connections should not pose any special difficulties in shipyard conditions, neither should the strength of the connection rely on very precise alignment.

and in relation to welding:

g. Welds must be accessible from both sides.

h. Flat (downhand) welds are easiest, overhead welds are difficult.

i. Intermittent welding is inadvisable except for very lightly loaded structure, because of the resulting stress concentrations and difficulty of preventing corrosion as well as causing local cyclic welding distortion.

j. Cruciform welds should be avoided as they are difficult to finish and are prone to cracking.

k. Butt welds are preferable to lap or fillet welds as they are more symmetrical, have better penetration and are easier to examine non-destructively.

l. The best designed structures have the minimum number of welds, with the maximum number of those being made using automatic processes.

m. For full details of recommended welding requirements for RN ships see NES706.

62 While on the theme of welded connections, most ships in modern shipyards are constructed in a modular manner, such that the hull is made up in a number of separate parts and welded together on completion. The result of this procedure is that there may well be a series of continuous butt connections running all round the hull, which could in adverse conditions constitute a weakness in relation to fatigue or brittle fracture. It is therefore of great importance to ensure that the material of the hull has adequate guaranteed toughness, as discussed in Chapter 5; however, even with tough hull material it is possible for the weld to create a weakness if it is not well made. If there is any doubt about the quality of the welds, that is if there has been limited or no non-destructive examination, or if the toughness of the hull material is in doubt, then it is essential that the butts in the stiffeners on the hull and strength decks are staggered away from the plate butts by at least 450 mm to provide some crack arrest capability. Alignment of the plate butts and the ends of the stiffeners is also important and should be in accordance with the design requirements, in the case of RN ships laid down in NES147.

Crack Arrestors

63 In rivetted ships any crack running through the structure was expected to be arrested at a rivetted joint. In welded construction there is no equivalent automatic crack arrestor and in early welded hulls some catastrophic failures resulted, and indeed unfortunately still do, see for example Corlett et al (1987). It therefore became normal in warships to fit crack arresting strakes at the upper deck edge and at the turn of bilge, which were originally rivetted. Later, with the development of notch tough steels, the rivetted strakes were replaced with welded strakes of tough material. The most reliable solution is clearly to build the whole hull of notch tough steel to crack arresting quality, that is to BS4360 grade 50EE or equivalent as indicated in Table 5.1 of

Chapter 5. This steel is a little more expensive than mild steel and a reasonable compromise is to construct the hull of steel with a moderate but guaranteed toughness (for example steel to BS4360 grade 43D or equivalent) and to fit tougher crack arrestors as before. It is not good practice to use a steel with no guaranteed toughness for the hull, even in the presence of crack arrestors, as the energy involved in a brittle failure may not be arrested in a single tough strake, and a highly embarrassing situation could arise even if the ship was in no immediate danger of being lost. Note that a minimum of four crack arresting strakes around the girth will always be necessary and it is also desirable to make the flat keel of crack arresting quality steel.

Design of Shaft Brackets

64 Shaft brackets may be of the A or P type but the same design principles apply in either case: the A bracket is more complex to design and so the method proposed here will concentrate on that shape. The cross section shape of the arms is defined from hydrodynamic considerations, and only a single dimension, either the cord length of the maximum thickness, from which all others are proportioned, is defined from strength considerations. The following traditional procedure therefore assumes that the arms are of equal length and are rigidly connected to the ship's structure, and that the cross sections of the two arms are identical and symmetrical about their longitudinal axes. If the arms are not of the same length then the method should be followed using the *longer* length of arm as this will cause the highest loads and the result will be conservative. If the arms are not rigidly attached to the hull, for example in a reinforced plastic vessel, more detailed calculations will be necessary, as discussed below.

65 A standard force F is assumed to act at the centre of the propeller at right angles to the shaft and to rotate with the shaft. The value of F used in the design of the shaft bracket will be the greatest of those derived from equations 12.5a, b or c. These expressions are taken from unpublished historical MOD work and relate respectively to the strength of the shaft, to the effect of loss of a single blade on the propeller and to the static weight of the propeller.

(12.5a) $$F_1 = 6.25\,\pi\sigma_{sy}\,(d_1^4 - d_2^4)/d_1^2$$

or

(12.5b) $$F_2 = 0.01 W ß N^2/g$$

or

(12.5c) $$F_3 = 9\ Wg$$

where σ_{sy} = the yield stress of the shaft material in MPa
d_1 = the outer diameter of the shaft in metres
d_2 = the inner diameter of the shaft in metres
W = the weight of the propeller in tonnes
N = the maximum propeller speed in RPM
ß = the maximum allowable propeller out of balance in N.m/tonne.

Note that with the units as stated F_1, F_2 and F_3 will all be in kN. F_2 is a first approximation to the force occurring when one blade of the propeller is lost, and is about 1000 times the out-of-balance force that might be allowed in normal service; if a better estimate is available of the allowable out of balance force due to a missing blade then it should be substituted for the one in equation 12.5b.

66 This rotating force F (the greatest of F_1, F_2 and F_3) together with a rotating moment proportional to Fh, where h is the distance between the transverse centre plane of the bracket and the plane of the propeller disc, is assumed to be mainly borne by the shaft bracket arms. The proportion of the moment which is carried by the bracket varies as the propeller rotates and a mean value of 0.8 Fh should be assumed.

67 The most important stresses in the arms which are set up by the rotating force and moment are due to longitudinal and transverse bending moments on the cross section, together with a uniform tension or compression. Other stresses are small and may be neglected. If the force F is acting in the plane of symmetry, that is the angle β to each arm as shown in *Figure 12.5*, the longitudinal bending moment M_1 due to the moment C (= 0.8 Fh) is approximately

$$M_1 = \pm 0.5\, C \sec \beta \tag{12.6}$$

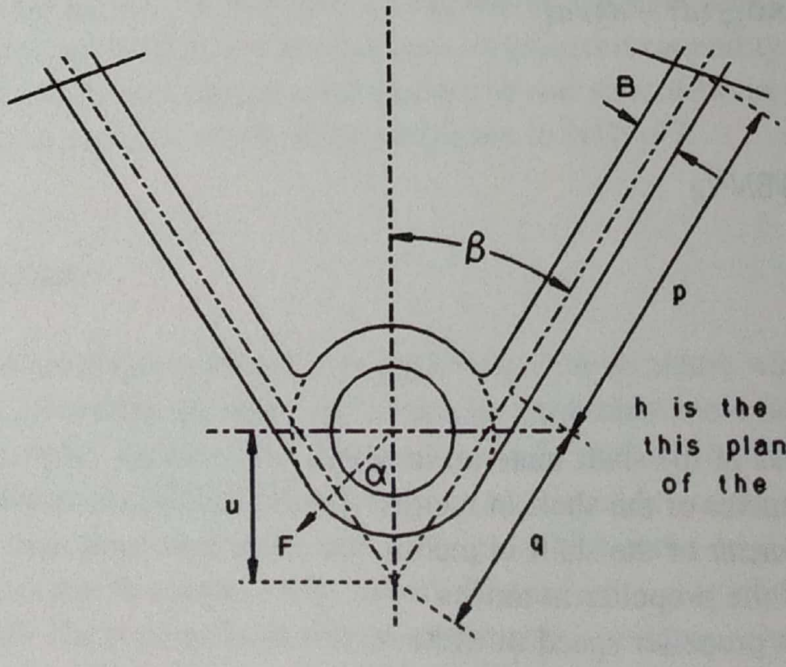
If F is acting at right angles to the plane of symmetry then the longitudinal bending moment M_2 on the arm is approximately

$$M_2 = \pm 0.5\, Q\, C \operatorname{cosec} \beta \tag{12.7}$$

where

$$Q = p\,(q/2 + p/3)/(q^2 + pq + p^2/3) \tag{12.7a}$$

with the parameters being defined in Figure 12.5.



h is the distance between this plane and the plane of the propeller disc.

Figure 12.5 Shaft Bracket Parameters

68 When F acts at an angle α to the plane of symmetry then the combined longitudinal bending moment M_{12} is

(12.8) $$M_{12} = M_1 \cos\alpha + M_2 \sin\alpha$$

with the signs of M_1 and M_2 depending on which arm is being considered. The maximum value of M_{12} is given by $\surd(M_1^2 + M_2^2)$ and occurs when tan α is equal to ±Q cot ß, the sign again depending on which arm is being considered.

69 The transverse bending moment M_3 has an approximate value given by

(12.9) $$M_3 = 0.5\, F\, Q\, u \sin\alpha$$

which reaches a maximum when α = 90° (u again is given in Figure 12.5). With radial arms u is zero and hence the transverse bending moment is also zero. However, as u reduces from its value for tangential arms the *longitudinal* stress tends to increase and its maximum may be greater than with the arms tangential. In general the difference between radial and tangential arms will not greatly affect the strength and the decision is best made from hydrodynamic or production considerations. If there are no good reasons against, tangential arms are usually the best compromise.

70 It will be noted from the above discussion that the maximum longitudinal and transverse bending moments occur at different values of α. The actual phase difference is a little more than ß but it is sufficiently accurate to take the phase angle as ß and the maximum combined stress is then that due to the maximum longitudinal bending moment acting simultaneously with the maximum value of the transverse bending moment multiplied by cos ß.

71 Section properties of a typical arm cross section with a cord length (c) to thickness ratio of 4 are as follows:

Area	$0.183\ c^2$
Centre of area from the leading edge	$0.479\ c$
Second moment of area about the centreline	$0.000673\ c^4$
Second moment of area about a transverse axis through the centre of area	$0.0106\ c^4$

and it is a relatively simple matter to calculate the properties of any other defined section. Knowing these values and the maximum combined bending moments on the arms, the bending stresses can be readily calculated for a chosen value of cord length.

72 Having combined the two bending stresses it is then necessary to subtract from the stress at the forward end of the cross section and add to that at the after end the stress due to the direct thrust whose maximum value is given by F/(A sin 2ß), where A is the cross section area of the arm. Note that any small angling of the arms away from the fore and aft direction to improve water flow may be taken as having a negligible effect on stresses.

73 The final stress thus obtained should not exceed the yield stress of the shaft bracket material, and no further margin is necessary, all the uncertainties being included with the estimation of load. The method described aims at achieving a design where the propeller and shaft will fail in an accident before the shaft bracket and so avoid unnecessary damage which could be very expensive to repair.

74 Where the arm enters the shell plating it may be necessary to crank the arm to achieve a reasonable internal arrangement for welding into the bottom stiffening. This cranking should be kept to a minimum, especially any change of direction which could cause a twisting moment to be applied and which the arm shape is not well designed to resist. Details of methods of attachment of shaft brackets to the hull are given for RN ships in NES155 Part 1.

75 For unusual bracket arrangements which are outside the assumptions stated in the method above, and particularly if non-metallic materials are involved in either the bracket or the hull, it will be necessary to use a more general analysis method. The ultimate would be a full three dimensional finite element analysis using solid elements to model the bracket, but in most cases it should be sufficient to treat the boss as rigid and the arms as beam elements, the main modelling problem being restricted to the material properties and the fixity into the hull.

Bilge Keels

76 Bilge keels are generally fitted to provide resistance to rolling, and so information on the preferred shape of bilge keels is given in hydrodynamics specifications. Bilge keels may also, under some circumstances, be used to structural effect in place of one or two hull longitudinal stiffeners. However, in this case it is important that the bilge keel is as extensive as possible and not broken up into short lengths, which usually happens when stabiliser fins are fitted. For most practical purposes therefore a bilge keel has little strengthening effect on the hull and so may be designed as a separate entity.

77 Bilge keels may be of either flat plate or hollow triangular construction. For the former, details are given in Lloyd's Rules, Part 3, Chapter 10 but it should be noted that for warships the use of a boundary bar (ground bar in Lloyd's parlance) is not considered appropriate due to the extra welding involved and the risk of creating further crack initiating points; the flat plate should be welded directly to the hull with a continuous full penetration fillet weld. Details of construction of triangular bilge keels are given for RN ships in NES155 Part 1 and again direct connection to the hull is preferred. In either type of bilge keel the plating thickness should be similar to that of the hull to which it is attached, but (for steel) not less than 15 mm for a flat plate or 10 mm for a triangular section. Note that the ends of the bilge keels will constitute stress concentrations and they should be positioned over hull stiffeners. If this is not possible then an insert plate should be fitted in the hull plate extending approximately 250 mm all round the end and about 50% thicker than the local hull plating; indeed, even if the keel is intended to end on a stiffener, it may be desirable to fit the insert because construction tolerances may be such that alignment with the stiffener cannot be guaranteed.

Rudders and Stabilisers

78 Information on the selection of shape and form of rudders and stabilisers is usually given in hydrodynamics specifications and from the details given therein the maximum forces and torques may be deduced. For rudders, further information may be found in Lloyd's Rules, Part 3, Chapter 13. In most modern warships spade rudders are fitted, but in some older vessels single pintle or 'gnomon' rudders may be found. Generally the forces, their points of action and the torques, together with the arrangement of bearings, will define the size and shape of the rudder stock. The construction of the blade, whatever the type of rudder, will usually consist of a fabricated framework on which the skin plating is attached. The framework should be designed as a grillage, neglecting the plating, in a form so as to transfer the loads adequately to the stock, but the spacing of the frame webs must be such that access is available for reliable construction and preservation. Having decided the arrangement of webs, Lloyd's Rules give a suitable method for sizing the members and the skin plating, but no part of the rudder should be less than 10 mm thick (in steel) to ensure adequate ruggedness and allowance for corrosion and erosion, which are particularly prevalent in such a structure which cannot be easily surveyed and preserved in service, and yet is of vital importance to ship safety.

79 Stabiliser blades are usually supplied as part of the stabiliser system and are unlikely to feature as part of the ship structural design.

Double Bottoms

80 During the design synthesis phase double bottom structures may be treated as a combination of outer bottom, internal deck and longitudinal bulkheads as described in Chapter 7. However, in the subsequent design and analysis phases such an approach is too simplistic because of the close integration and complexity of the structure. Work undertaken at Cardiff University and reported by Evans et al (1984) provides a means for analysing double bottom structures under various load combinations as if the whole structure was a deep grillage.

Axial Loads

81 Under axial loads it is sufficient to treat the vertical members (the longitudinal and transverse plate floors) as the webs in a grillage with the plating of the inner and outer bottom acting as flanges. The effective width of plating to be used is discussed at length by Evans et al (1984) but for practical purposes half the web spacing is adequate and will be a little conservative for lower values of plate slenderness ß.

82 The critical buckling stress of the webs in compression can be found using the appropriate column collapse load shortening curves (data sheets 18.9 to 18.14) but in this case the value of ß should be for the full web spacing and thickness of the *outer* bottom while the value of λ should only include the *effective* widths of the inner and outer bottom. The span to be used for λ should be the spacing of the webs at right angles to the applied load.

83 Overall buckling is unlikely to be a critical mode unless the double bottom is very shallow. If there is any doubt, the procedure discussed in Chapter 7 using data sheets 18.15 to 18.29 may be used.

Lateral Loads

84 For design of a double bottom under lateral load the procedure of section 9 of BS5400 Part 3 for steel box girder bridges is adequate. This is a comprehensive and fairly complex process and it is, therefore, not suitable for reproduction in this book.

Combined Loads

85 A full description of the interaction between axial and lateral loads is presented by Evans et al (1984). For practical purposes, however, and slightly conservatively, the interaction diagram may be taken to be a quadrant of a circle, that is

$$(L/L_u)^2 + (P/P_u)^2 = 1 \tag{12.10}$$

where L and P are the applied axial load and lateral pressure respectively, and L_u and P_u are the equivalent strengths for the load applied independently.